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TRANSLATION

EXPERIMENTAL RESEARCH OF THE HYDRODYNAMICS OF A TURBULENT FLOW OF AIR IN THE GAP BETWEEN ROTATING COAXIAL CYLINDERS

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FOREIGN TECHNOLOGY DIVISION

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EXPERIMENTAL RESEARCH OF THE HYDRODYNAMICS OF A TURBULENT FLOW OF AIR IN THE GAP BETWEEN ROTATING COAXIAL CYLINDERS

by

S. I. Kosterin, Yu. A. Koshmarov, and Yu. P. Finat'yev

The article tells about the results of experimental researches in the flow of air in the ring channel between a rotating inner cylinder and a stationary outer one. There are established boundaries of the regions of flow for the various systems. The results are presented of the experimental measurements of the losses by friction.

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The most widely used figures for the relative magnitudes of the ring gaps in the practice of aerodynamic designing (electrical machines, generators, etc.) lie within the limits $(r_2 - r_1)/r_1 = 0$--0.3 (here r_1 is the radius of the rotor; r_2 the radius of the stator).

At the present time there are being conducted a series of experimental and theoretical researches in hydrodynamics of the purely turbulent flow in the ring gaps the relative magnitude of which is close to zero [1, 2]. For gaps, however, of medium and large relative magnitude experimental and theoretical investigations of the hydrodynamics and heat exchange in a turbulent flow are hardly being conducted at all.

In this study we will report on results from experimental investigations of the turbulent flow in a ring channel the relative magnitude of which amounts to $(r_2 - r_1)r_1 = 0.271$.

Experimental Unit and Methods of Measurements

Experimental investigations were conducted with an arrangement which consisted of two coaxial cylinders. The outer cylinder was stationary; the inner one rotated. The diameter of the rotating cylinder (rotor) was 192 mm;

the inner diameter of the stator was 244 mm The size of the ring gap was $\delta = r_2 - r_1 = 26$ mm. The length of the channel was 2,015 mm. The working surface of the outer cylinder (stator) after the lathe work was polished. The relative roughness of the working surface of the rotor amounted to $r_1/k \approx 10{,}000$—15,000. For assuring the uniform input of air into the gap there was provided in the input nipple a device which liquidated the after-effect of the radial introduction of the air to the working area. Measurements of the profiles of the velocities at the input of the air into the channel showed that the distribution of the velocities in the input section was always practically uniform.

On the stator at every 100 mm there were drilled holes of the diameter of 1 mm for measuring the static pressure along the channel. For measuring the profiles and pulsations of the velocities on the stator there were set up in some sections corresponding probes the displacement of which was fixed on a micrometric scale. For measuring the velocities there were used tubes of full pressure of the diameter of 0.8 mm. The measurement of the pulsations of the velocity by a thermo anemometer of the type ETAM-3A.

For measuring the tangential stress of the friction against the wall of the stator at a section distant from the entrance into the channel by 1 m. there was built in a pickup of the friction with a floating element prepared in the laboratory on the basis of torsion weights of the brand VT-200 with gradations of 0.2 mg. As an operating force there was used air compressed by a piston compressor, which was previously dried and cleaned. In the scheme of supplying the air, for leveling off the fluctuations in the pressure, there was a buffer cavity. The greatest use of air in the experiments amounted to 30 kg/sec.

The flow diaphragm was placed immediately in front of the working area.

The diameter of the opening in the plate was 45 mm; the diameter of the tube 82 mm. The diaphragm was designed in accordance with the norms for the measurement of the flow of liquids, gas, and steam [3]. The consumption of air was determined on the basis of the data of these norms. The results of the measurements of the use of air by this diaphragm were compared with those computed by the formulas of the velocities. The two methods satisfactorily agreed.

For the study of the effect of heat exchange on the character of the flow of air in a ring channel there was prepared a second variant of the rotor with exactly the same diameter as the first. Inside this rotor there was located a uniformly wound spiral of nichrome tape, the end of which were brought out onto a slip ring. The rings of the slip-ring device were made of copper; the brushes of copper graphite. The power of the electric preheater of the rotor was regulated. For measuring the temperature of the surface of the rotor and the stator into the walls of these there were built chromel-copel thermocouples. The thermocouples from the rotor are brought out through a slip ring.

Systems of Flow in the Ring Channel

Experimental investigations [4] revealed the existence of four systems of flow of the fluid in the ring channels. At the same time investigation was made of the character of flow of a fluid in ring gaps the relative magnitude of which amounted to 0.214 and 0.362 in a range of variation of the Reynolds number $Re_{za} = V_{za}(r_2 - r_1)/\nu = 0 - 10^3$ and Taylor's number

$$\mathrm{Te} = \omega_1 \left(\frac{r_2 + r_1}{2}\right)^{1/2} (r_2 - r_1)^{3/2}/\nu = 0 - 700. \qquad \text{(a)}$$

Most of the technical problems are characterized by higher values for the numbers Re_{za} and Te.

Our researches conducted with the same relative ring gaps enabled us to study the character of the flow of a fluid in the range of changes of the indicated criteria: $Re_{za} = 10^3 - 10^4$; $Te = 0 - 5 \cdot 10^4$. The procedure of the experiment consisted in the following. With an unchanging feed of air there was a gradual increase in the rpm of the rotor. For each of its figures measurements were made of the profiles of the velocities, the distribution of static pressures along the wall of the outer cylinder, and the character of the pulsations of the velocities. In studying the character of the motion in non-isothermic conditions there were measured the temperatures of the wall of the preheated rotor, and of the wall of the stator and of the air. With the change in the system of flow there was a sharp change in the character of the distribution of the velocities and pressures along the walls of the stator.

With a laminar motion pulsations of the velocity are absent; with a turbulent motion they have an irregular character. Under the turbulent system with vortices there arise regular pulsations of velocity with great amplitude, on which are superimposed irregular pulsations of little amplitude. The regular pulsations of velocity correspond to the stable, large vortices which arise in this system of flow, which are commensurate with the dimensions of the gap. This was confirmed visually by observations made in [4]. The character of the distribution of the velocities also noticeably changes with the coming of a new system. Especially noticeable is the transformation of the profiles of the tangential velocities. Where there is purely turbulent motion the tangential velocity of the air constantly becomes less along the radius of the rotor to the stator. The system of turbulent motion with vortices is characterized by nonmonotone change in the tangential velocity along the radius.

The drop in the static pressures along the wall of the stator under

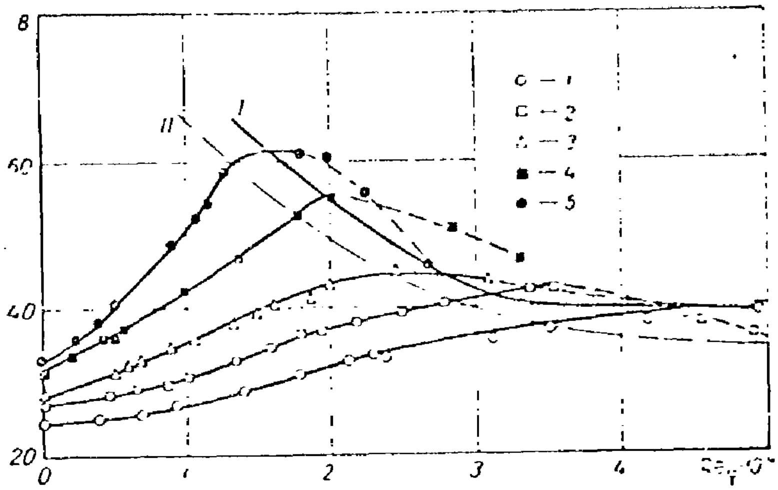

Fig. 1. Distribution of the static pressures $\frac{\Delta p}{\Delta z} \frac{4(r_2 - r_1)}{\rho v_{za}^2} = B$ along the wall of the inner cylinder (stator) with different rpm $Re_\varphi = \omega_1 r_1 (r_2 - r_1)/\nu$ in different amounts of air used Re_{za}: 1 — $1.5 \cdot 10^4$; 2 — $1.1 \cdot 10^4$; 3 — $8.8 \cdot 10^3$; 4 — $5.08 \cdot 10^3$; 5 — $4.5 \cdot 10^3$; I and II, respectively, equal the curves drawn through the points in which

$$\frac{\partial B}{\partial Re_\varphi} = 0 \text{ и } \frac{\partial^2 B}{\partial Re_\varphi^2} = 0 \qquad \text{(b)}$$

The drop in the static pressures along the wall of the stator during turbulent motion constantly increased with the increase in the rpm. After the ensuance of the system of flow with vortices a reverse picture was observed. The distribution of static pressures along the wall of the stator on the basic (working) area of the channel under the turbulent motion became linear. With the onset of the system with vortices also there was maintained an approximately linear distribution. In Fig. 1 there are shown the results of these measurements for a number of values of the Reynolds number of the axial motion. By joining the points corresponding to the condition $\frac{\partial}{\partial Re_\varphi}\left[\frac{\Delta p}{\Delta z} \frac{4(r_2 - r_1)}{\rho v_{za}^2}\right] = 0$ (extremum) we will get a line which separates the field of the graph into two areas—of purely turbulent flow and turbulent motion with vortices. For more precise determination of the boundaries of the transition, it is clear, one should use the condition of the equality zero of the second derivative

$$\frac{\partial^2}{\partial \mathrm{Re}_\varphi^2}\left[\frac{\Delta p}{\Delta z}\,\frac{4(r_2-r_1)}{\rho v_{za}^2}\right]=0. \qquad \text{(c)}$$

The boundary drawn along these points will determine some large area for the flow with vortices. The points of transition obtained by this method find themselves in complete correspondence with the data on the moment of changeover of the character of the profile of the velocities and pulsation velocities.

The investigations of the systems of flow under nonisothermic conditions showed that the presence of an insignificant heat exchange has little effect on the point of transmition. As depends on the numbers Re_{za} and Te there may exist laminar flow, laminar flow with vortices, purely turbulent flow, and turbulent motion with large-scale retular vortices.

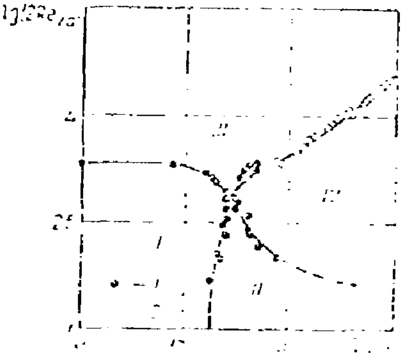

Fig. 2. Systems of flow in a ring channel between rotating cylinders

$$\mathrm{Te}=\left[\omega_1\left(\frac{r_2+r_1}{2}\right)^{1/2}(r_2-r_1)^{3/2}\right]\Big/\nu; \qquad \text{(d)}$$

$$\mathrm{Re}_{za}=\frac{v_{za}(r_2-r_1)}{\nu};$$

I, area of laminar flow; II, of laminar flow with vortices; III, of purely turbulent flow; IV, of turbulent flow with vortices; 1, 2, respectively experimental data [4] and ours.

In the range of numbers $\mathrm{Re}_{za} = 10^3 - 10^4$ and $\mathrm{Te} = 10^3 - 10^4$ the boundary between the system of purely turbulent and turbulent motion with large-scale vortices can be approximately determined by the formula obtained on the basis of the treatment given in Fig. 2 in the following form:

$$\mathrm{Te}_{\kappa p}\approx 0{,}015\,\mathrm{Re}_{za}^{1,5}. \qquad (1)$$

The applicability of this formula for other gaps requires experimental confirmation.

Results of the Measurements of the Friction of the Air Against the Wall of the Ring Gap

In considering the process of the movement of the air in the axial direction one can separate the channel into two zones—the zone corresponding to the motion $\left(v_r = 0, \frac{\partial v_z}{\partial z} = 0, \frac{\partial v_\varphi}{\partial z} = 0\right)$ and the zone of unsettled motion, the beginning area. The length of the beginning area in our researches as the measurements showed under all regimes did not exceed the value 0.3—0.7 m. As the working area of the channel one considers that part of it included between the sections removed from the inpu by 0.7 m and from the output by 0.4 m. In the experiments there was studied only the settled movement of the air. In the analysis of the losses of pressure along the channel there were used data obtained in the working area of the channel. In the middle part of the working channel on the stator there was installed a pickup of the tangential friction. The profiles of the velocities were measured along the whole length of the channel at every 300 mm, which made it possible to follow up on the transformation of the profiles in the beginning area and reveal the zone of settled flow.

For the area of settled motion under the purely turbulent system of flow of air the figure for the coefficient of the axial friction was computed in accordance with the formula

$$\lambda_z = 4\,\frac{\Delta p}{\Delta z}\,\frac{(r_2 - r_1)}{\rho v_{za}^2}\,. \qquad (2)$$

The data obtained for the coefficients of the axial friction in the range $Re_{za} = 10^3 - 10^4$ with absence of rotation of the inner cylinder corresponds well with Blasius's formula

$$\lambda_{z0} = \frac{0{,}3164}{Re_{za}^{0.25}}\,. \qquad (3)$$

where $Re_{za} = \left[2v_{za}\,(r_2 - r_1)\right]/\nu$.

In the case of the purely turbulent system of the flow and with a constant number Re_{za} (constant consumption of air) one observes an increase in the coefficient of resistance with the increase in the number Re_{φ}, i. e., with the increase in the rpm.

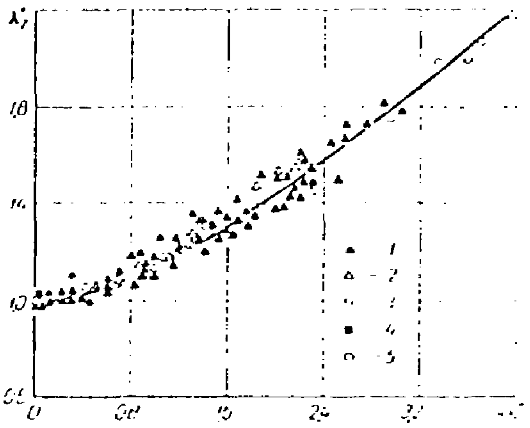

Fig.3 Dependence of $\lambda_z^* = \lambda_z/\lambda_{zo}$ on $Re_{\varphi}/Re_{za} = Re^*$, experimental data: 1, ours; 2 and 3, [1]; 4, [5]; 5, [6]; solid lines is the theoretical solution [3].

The results of the experimental measurements of the coefficient of the hydraulic losses of the axial movement for the purely turbulent system of flow (Fig. 3) were presented in the form of a function

$$\frac{\lambda_z}{\lambda_{z0}} = f\left(\frac{Re_{\varphi}}{Re_{za}}\right). \quad (4)$$

The value λ_{zo} in this case was determined by the formula (3). The results of our experiments and researches [1] agree well with each other and with the theoretical data:

$$\lambda_z = \lambda_{z0}(1 + 2Z_a^2)^{0.35}, \quad (5)$$

where

$$Z_a = \frac{1}{2}\,\frac{Re_{\varphi}}{Re_{za}}. \quad (6)$$

In this way the computation of the hydraulic losses for the axial motion in a purely turbulent system of flow in the range of numbers $Re_{za} = 10^3 - 10^4$ can be carried out by making use of the formula (5) for gaps the relative size of which lies within the limits 0 to 0.27. The magnitude $4\frac{\Delta}{\Delta}\frac{p}{z}\left(\frac{r - r}{\rho v^2_{za}}\right)$ computed in accordance with the distribution of the measured static pressures on the wall of the stator for the area of flow with vortices apparently does not complete characterize the hydraulic losses by friction connected with the axial motion of the fluid.

Along with the measurements of the hydraulic losses of the axial motion measurements were made of the stress of the tangential friction on the wall of the stator. From the condition of constancy of moment of friction for the working area the figure for the stress of friction on the rotor can be determined in accordance with the formula

$$\tau_{\varphi 1} = \tau_{\varphi 2}\left(\frac{r_2}{r_1}\right)^2. \qquad (7)$$

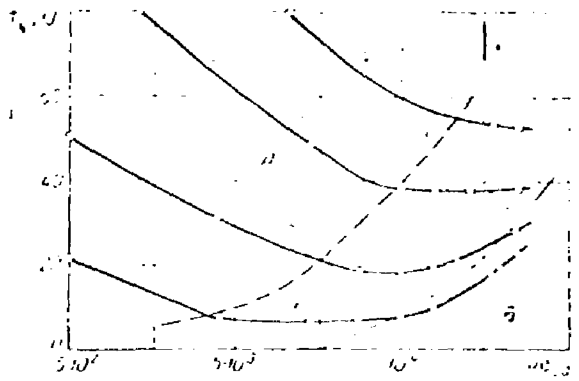

Fig. 4. Dependence of the tangential friction $\tau_{\varphi 2}$ (kΓ /m^2) on the parameter Re_{za} under varying consumption of air and different figures for the rpm. 1, 800 rpm; 2, 1,200; 3, 1,700; 4, 2,000 rpm; A, zone of turbulent flow with vortices; Б, zone of purely turbulent flow; I, boundary of the systems found from the condition

$$\frac{\partial}{\partial Re_\varphi}\left[\frac{\Delta p}{\Delta z}\,\frac{4(r_2-r_1)}{\rho v_{za}^2}\right]=0; \qquad (8)$$

II, from the condition

$$\frac{\partial^2}{\partial Re_\varphi^2}\left[\frac{\Delta p}{\Delta z}\,\frac{4(r_2-r_1)}{\rho v_{za}^2}\right]=0. \qquad (9)$$

The sensitivity of the pick-up on the one hand and the limitedness of the amount of air compressible by the compressors on the other, made it possible to make the measurements of the stress of friction only within a narrow zone of measurements of the numbers Re_z and Re_φ, corresponding to the area of purely turbulent motion. The error in the measurements could amount to 20 or 30%. The results of the measurements with different figures for the rpm of the rotor are given in Fig. 4 in the form of dependence of the stress of tangential friction on the stator $\tau_{\varphi 2}$ on the Reynolds number of the axial motion (consumption. In this same graph there are plotted the boundaries of the areas of friction.

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